IN THE SPECIFICATION:

Please amend the first four full paragraphs appearing on page 1, lines 4-17 as follows:

This application claims ~~the benefit of~~ priority to U.S. Provisional Patent Application Serial No. 60/421,737, filed October 28, 2002, which is incorporated herein by reference.

This application also claims priority to U.S. Provisional Patent Application Serial No. 60/514,670, titled "HEAT TRANSFER SYSTEM FOR A CYCLICAL HEAT EXCHANGE SYSTEM," filed ~~on~~ October 28, 2003, which also is incorporated herein by reference.

This application is a continuation-in-part of U.S. Patent Application Serial No. 10/676,265, titled "EVAPORATOR FOR A HEAT TRANSFER SYSTEM," filed October 2, 2003, which claimed priority to U.S. Patent Application Serial No. 60/415,424, filed October 2, 2003, which are ~~alos~~ also incorporated herein by reference.

This application is a continuation-in-part of U.S. ~~application Ser.~~ Patent Application Serial No. 10/602,022, filed June 24, 2003, now U.S. Patent 7,004,240, issued February 28, 2006, which claims the benefit of U.S. Provisional Patent Application Serial No. 60/391,006, filed June 24, 2002 and is a continuation-in-part of U.S. ~~application Ser.~~ Patent Application Serial No. 09/896,561, filed June 29, 2001, now U.S. Patent 6,889,754, issued May 10, 2005, which claims the benefit of U.S. Provisional Patent Application Serial No. 60/215,588, filed June 30, 2000. All of these applications are incorporated herein by reference.

Please amend the sixth full paragraph appearing on page 1, lines 24-29, as follows:

Heat transfer systems are used to transport heat from one location (the heat source) to another location (the heat sink). Heat transfer systems can be used in terrestrial or extraterrestrial applications. For example, heat transfer systems may be integrated by satellite equipment that operates within ~~zero~~ zero- or low-gravity environments. As another example, heat transfer systems can be used in electronic equipment, which often requires cooling during operation.

Please amend the paragraph bridging pages 1 and 2 (page 1, line 30, through page 2, line 10), as follows:

Loop Heat Pipes (LHPs) and Capillary Pumped Loops (CPLs) are passive two-phase heat transfer systems. Each includes an evaporator thermally coupled to the heat source, a condenser thermally coupled to the heat sink, fluid that flows between the evaporator and the condenser, and a fluid reservoir for expansion of the fluid. The fluid within the heat transfer system can be referred to as the working fluid. The evaporator includes a primary wick and a core that includes a fluid flow passage. Heat acquired by the evaporator is transported to and discharged by the condenser. These systems utilize capillary pressure developed in a ~~fine-pored~~ fine-pored wick within the evaporator to promote circulation of working fluid from the evaporator to the condenser and back to the evaporator. The primary distinguishing characteristic between an LHP and a CPL is the location of the loop's reservoir, which is used to store excess fluid displaced from the loop during operation. In general, the reservoir of a CPL is located remotely from the evaporator, while the reservoir of an LHP is co-located with the evaporator.

Please amend the third full paragraph appearing on page 2, lines 22-28, as follows:

The evaporator includes a liquid barrier wall containing the working fluid on an inner side of the liquid barrier wall, which working fluid flows only along the inner side of the liquid barrier wall, wherein the primary wick is positioned between ~~the~~ a heated wall and the inner side of the liquid barrier wall; a vapor removal channel that is located at an interface between the primary wick and the heated wall, the vapor removal channel extending to a vapor outlet; and a liquid flow channel located between the liquid barrier wall and the primary wick, the liquid flow channel receiving liquid from a liquid inlet.

Please amend the second full paragraph appearing on page 4, lines 11-14, as follows:

Integral incorporation of the evaporator or condenser with the heat ~~sourceof~~ source of the cyclical heat exchange system can minimize packaging size. On the other hand, if the evaporator

or condenser is clamped onto the heat source, the deployment and replacement of parts is facilitated.

Please amend the fifth full paragraph appearing on page 4, lines 23-26, as follows:

The heat transfer system combines efficient heat transfer mechanisms (evaporation and condensation) to couple the fluid of the Stirling cycle (helium) to the ultimate heat sink (ambient air). Consequently, a significant improvement in Stirling ~~Cycle~~ cycle efficiency (for example, up to 50%) is provided.

Please amend the third full paragraph appearing on page 5, lines 11-13, as follows:

A loop heat pipe of the present invention provides for efficient packaging with a cylindrical refrigerator by adapting the traditional cylindrical geometry of a LHP evaporator to a planar ~~"flat-plate "~~ "flat-plate" geometry that can be wrapped in an annular shape.

Please amend the first full paragraph appearing on page 8, lines 1-2, as follows:

FIG. 25 is a schematic diagram of a heat transfer system using an evaporator designed in accordance with the principles of FIGS. ~~10-13.~~ 11-13.

Please amend the third full paragraph appearing on page 8, lines 4-5, as follows:

FIG. ~~27is~~ 27 is a partial cross-sectional detail view of an evaporator used in the heat transfer system of FIG. 25.

Please amend the seventh and eighth full paragraphs appearing on page 8, lines 13-16, as follows:

FIG. 31 is a partial cross-sectional elevation view (taken along line ~~31-31)of~~ 31-31) of the heat transfer system packaged around the cyclical heat exchange system portion of FIG. 30.

FIG. 32 is a partial cross-sectional elevation view (taken at detail ~~3200)of~~ 3200) of the interface between the heat transfer system and the cyclical heat exchange system of FIG. 30.

Please amend the seventh full paragraph appearing on page 9, lines 14-22, as follows:

As discussed above, in a loop heat pipe (LHP), the reservoir is co-located with the evaporator, thus, the reservoir is thermally and hydraulically connected with the reservoir through a heat-pipe-like conduit. In this way, liquid from the reservoir can be pumped to the evaporator, thus ensuring that the primary wick of the evaporator is sufficiently wetted or "primed" during start-up. Additionally, the design of the LHP also reduces depletion of liquid from the primary wick of the evaporator during steady-state or transient operation of the evaporator within a heat transport system. Moreover, vapor and/or bubbles of ~~non-condensable~~ non-condensable gas (NCG bubbles) vent from a core of the evaporator through the heat-pipe-like conduit into the reservoir.

Please amend the third full paragraph appearing on page 10, lines 12-21, as follows:

The heat transfer system 105 includes a main evaporator 115, and a condenser 120 coupled to the main evaporator 115 by a liquid line 125 and a vapor line 130. The condenser 120 is in thermal communication with a heat sink 165, and the main evaporator 115 is in thermal communication with a heat source ~~Qin~~ Q_{in} 116. The heat transfer system 105 may also include a hot reservoir 147 coupled to the vapor line 130 for additional pressure containment, as needed. In particular, the hot reservoir 147 increases the volume of the heat transport system 100. If the working fluid is at a temperature above its critical temperature, that is, the highest temperature at which the working fluid can exhibit liquid-vapor equilibrium, its pressure is proportional to the mass in the heat transport system 100 (the charge) and inversely proportional to the volume of the heat transport system 100. Increasing the volume with the hot reservoir 147 lowers the fill pressure.

Please amend the paragraph bridging pages 10 and 11 (page 10, line 22, through page 11, line 2), as follows:

The main evaporator 115 includes a container 117 that houses a primary wick 140 within which a core 135 is defined. The main evaporator 115 includes a bayonet tube 142 and a secondary wick 145 within the core 135. The bayonet tube 142, the primary wick 140, and the

secondary wick 145 define a liquid passage 143, a first vapor passage 144, and a second vapor passage 146. The secondary wick 145 provides phase control, that is, liquid/vapor separation in the core 135, as discussed in U.S. ~~application Ser.~~ Patent Application Serial No. 09/896,561, filed June 29, 2001, now U.S. Patent No. 6,889,754, issued May 10, 2005, which is incorporated herein by reference in its entirety. As shown, the main evaporator 115 has three ports, a liquid inlet 137 into the liquid passage 143, a vapor outlet 132 into the vapor line 130 from the second vapor passage 146, and a fluid outlet 139 from the liquid passage 143 (and possibly the first vapor passage 144, as discussed below). Further details on the structure of a three-port evaporator are discussed below with respect to FIGS. 5A and 5B.

Please amend the first and second full paragraphs appearing on page 11, lines 3-24, as follows:

The priming system 110 includes a secondary or priming evaporator 150 coupled to the vapor line 130 and a reservoir 155 co-located with the secondary evaporator 150. The reservoir 155 is coupled to the core 135 of the main evaporator 115 by a secondary fluid line 160 and a secondary condenser 122. The secondary fluid line 160 couples to the fluid outlet 139 of the main evaporator 115. The priming system 110 also includes a controlled heat source ~~Qsp~~ Q_{sp} 151 in thermal communication with the secondary evaporator 150.

The secondary evaporator 150 includes a container 152 that houses a primary wick 190 within which a core 185 is defined. The secondary evaporator 150 includes a bayonet tube 153 and a secondary wick 180 that ~~extend~~ extends from the core 185, through a conduit 175, and into the reservoir 155. The secondary wick 180 provides a capillary link between the reservoir 155 and the secondary evaporator 150. The bayonet tube 153, the primary wick 190, and the secondary wick 180 define a liquid passage 182 coupled to the secondary fluid line 160, a first vapor passage 181 coupled to the reservoir 155, and a second vapor passage 183 coupled to the vapor line 130. The reservoir 155 is thermally and hydraulically coupled to the core 185 of the secondary evaporator 150 through the liquid passage 182, the secondary wick 180, and the first vapor passage 181. Vapor and/or NCG bubbles from the core 185 of the secondary evaporator 150 are swept through the first vapor passage 181 to the reservoir 155 and

condensable liquid is returned to the secondary evaporator 150 through the secondary wick 180 from the reservoir 155. The primary wick 190 hydraulically links liquid within the core 185 of the secondary evaporator 150 to the controlled heat source ~~Qsp~~ Q_{sp} 151, permitting liquid at an outer surface of the primary wick 190 to evaporate and form vapor within the second vapor passage 183 when heat is applied to the secondary evaporator 150.

Please amend the third full paragraph appearing on page 12, lines 9-13, as follows:

Referring also to FIG. 3, the heat transport system 100 performs a procedure 300 for transporting heat from the heat source ~~Qin~~ Q_{in} 116 and for ensuring that the main evaporator 115 is wetted with liquid prior to startup. The procedure 300 is particularly useful when the heat transfer system 105 is at a supercritical state. Prior to initiation of the procedure 300, the heat transport system 100 is filled with a working fluid at a particular pressure, referred to as a "fill pressure."

Please amend the paragraph bridging pages 12 and 13 (page 12, line 23, through page 13, line 7), as follows:

Meanwhile, power is applied to the priming system 110 by applying heat from the heat source ~~Qsp~~ Q_{sp} 151 to the secondary evaporator 150 (step 315) to enhance or initiate circulation of fluid within the heat transfer system 105. Vapor output by the secondary evaporator 150 is pumped through the vapor line 130 and through the condenser 120 (step 320) due to capillary pressure at the interface between the primary wick 190 and the second vapor passage 183. As vapor reaches the condenser 120, it is converted to liquid (step 325). The liquid formed in the condenser 120 is pumped to the main evaporator 115 of the heat transfer system 105 (step 330). When the main evaporator 115 is at a higher temperature than the critical temperature of the fluid, the liquid entering the main evaporator 115 evaporates and cools the main evaporator 115. This process (steps 315-330) continues, causing the main evaporator 115 to reach a set point temperature (step 335), at which point the main evaporator 115 is able to retain liquid and be wetted and to operate as a capillary pump. In one implementation, the set point temperature is the temperature to which the reservoir 155 has been cooled. In another implementation, the set

point temperature is a temperature below the critical temperature of the working fluid. In a further implementation, the set point temperature is a temperature above the temperature to which the reservoir 155 has been cooled.

Please amend the first full paragraph appearing on page 13, lines 8-23, as follows:

If the set point temperature has been reached (step 335), the heat transport system 100 operates in a main mode (step 340) in which heat from the heat source ~~Qin~~ Q_{in} 116 that is applied to the main evaporator 115 is transferred by the heat transfer system 105. Specifically, in the main mode, the main evaporator 115 develops capillary pumping to promote circulation of the working fluid through the heat transfer system 105. Also, in the main mode, the set point temperature of the reservoir 155 is reduced. The rate at which the heat transfer system 105 cools down during the main mode depends on the ~~cold biasing~~ cold-biasing of the reservoir 155 because the temperature of the main evaporator 115 closely follows the temperature of the reservoir 155. Additionally, though not required, a heater can be used to further control or regulate the temperature of the reservoir 155 during the main mode (step 340). Furthermore, in the main mode, the power applied to the secondary evaporator 150 by the controlled heat source ~~Qsp~~ Q_{sp} 151 is reduced, thus bringing the heat transfer system 105 down to a normal operating temperature for the fluid. For example, in the main mode, the heat load from the controlled heat source ~~Qsp~~ Q_{sp} 151 to the secondary evaporator 150 is kept at a value equal to or in excess of heat conditions, as defined below. In one implementation, the heat load from the controlled heat source ~~Qsp~~ Q_{sp} is kept to about 5 to 10% of the heat load applied to the main evaporator 115 from the heat source ~~Qin~~ Q_{in} 116.

Please amend the first and second full paragraphs appearing on page 14, lines 5-29, as follows:

To reduce the adverse impact of heat conditions discussed above, the priming system 110 operates at a power level ~~Qsp 151~~ greater than or equal to the sum of the head conduction and the parasitic heat gains. As mentioned above, for example, the priming system 110 can operate at ~~5-10%~~ 5 to 10% of the power to the heat transfer system 105. In particular, fluid that includes a

combination of vapor bubbles and liquid is swept out of the core 135 for discharge into the secondary fluid line 160 leading to the secondary condenser 122. In particular, vapor that forms within the core 135 travels around the bayonet tube ~~143~~ 142 directly into the fluid outlet port 139. Vapor that forms within the first vapor passage 144 makes it way into the fluid outlet port 139 by either traveling through the secondary wick 145 (if the pore size of the secondary wick 145 is large enough to accommodate vapor bubbles) or through an opening at an end of the secondary wick 145 near the outlet port 139 that provides a clear passage from the first vapor ~~passages~~ passage 144 to the outlet port 139. The secondary condenser 122 condenses the bubbles in the fluid and pushes the fluid to the reservoir 155 for reintroduction into the heat transfer system 105.

Similarly, to reduce parasitic heat input to the liquid line 125, the secondary fluid line 160 and the liquid line 125 can form a coaxial configuration and the secondary fluid line 160 surrounds and insulates the liquid line 125 from surrounding heat. This implementation is discussed further below with reference to FIGS. 8A and 8B. As a consequence of this configuration, it is possible for the surrounding heat to cause vapor bubbles to form in the secondary fluid line 160, instead of in the liquid line 125. As discussed, by virtue of capillary action ~~affected~~ effected at the secondary wick 145, fluid flows from the main evaporator 115 to the secondary condenser 122. This fluid flow, and the relatively low temperature of the secondary condenser 122, causes a sweeping of the vapor bubbles within the secondary fluid line 160 through the secondary condenser 122, where they are condensed into liquid and pumped into the reservoir 155.

Please amend the paragraph bridging pages 14 and 15 (page 14, line 30, through page 15, line 12), as follows:

As shown in FIG. 4, data from a test run is shown. In this implementation, prior to startup of the main evaporator 115 at ~~temperature~~ time 410, a temperature 400 of the main evaporator 115 is significantly higher than a temperature 405 of the reservoir 155, which has been cold-biased to the set point temperature (step 305). As the priming system 110 is wetted (step 310), power ~~Qsp~~ Q_{sp} 450 is applied to the secondary evaporator 150 (step 315) at a

time 452, causing liquid to be pumped to the main evaporator 115 (step 330), the temperature 400 of the main evaporator 115 drops until it reaches the temperature 405 of the reservoir 155 at time 410. Power ~~Qin~~ Q_{in} 460 is applied to the main evaporator 115 at a time 462, when the system 100 is operating in LHP mode (step 340). As shown, power input ~~Qin~~ Q_{in} 460 to the main evaporator 115 is held relatively low while the main evaporator 115 is cooling down. Also shown are the temperatures 470 and 475, respectively, of the secondary fluid line 160 and the liquid line 125. After time 410, temperatures 470 and 475 track the temperature 400 of the main evaporator 115. Moreover, a temperature 415 of the secondary evaporator 150 follows closely with the temperature 405 of the reservoir 155 because of the thermal communication between the secondary evaporator 150 and the reservoir 155.

Please amend the first full paragraph appearing on page 15, lines 13-30, as follows:

As mentioned, in one implementation, ethane may be used as the fluid in the heat transfer system 105. Although the critical temperature of ethane is 33°C, for the reasons generally described above, the heat transport system 100 can start up from a supercritical state in which the heat transport system 100 is at a temperature of 70°C. As power ~~Qsp~~ Q_{sp} 450 is applied to the secondary evaporator 150, the temperatures of the condenser 120 and the reservoir 155 drop rapidly (between times 452 and 410). A trim heater can be used to control the temperature of the reservoir 155 and thus the condenser 120 ~~to~~ operates at a temperature of -10°C. To startup the main evaporator 115 from the supercritical temperature of 70°C, a heat load or power input ~~Qsp~~ Q_{sp} of ~~10W~~ 10 W is applied to the secondary evaporator 150. Once the main evaporator 115 is primed, the power input from the controlled heat source ~~Qsp~~ Q_{sp} 151 to the secondary evaporator 150 and the power applied to and through the trim heater both may be reduced to bring the temperature of the heat transport system 100 down to a nominal operating temperature of about -50°C. For instance, during the main mode, if a power input ~~Qin~~ Q_{in} of ~~40W~~ 40 W is applied to the main evaporator 115, the power input ~~Qsp~~ Q_{sp} to the secondary evaporator 150 can be reduced to approximately ~~3W~~ 3 W while operating at -45°C to mitigate the 3W lost through heat conditions (as discussed above). As another example, the main evaporator 115 can operate with power input ~~Qin~~ Q_{in} from about ~~10W~~ 10 W to about ~~40W~~ 40 W with ~~5W~~ 5 W

applied to the secondary evaporator 150 and with the temperature 405 of the reservoir 155 at approximately -45°C.

Please amend the paragraph bridging pages 15 and 16 (page 15, line 31, through page 16, line 19), as follows:

Referring to FIGS. 5A and 5B, in one implementation, the main evaporator 115 is designed as a three-port evaporator 500 (which is the design shown in FIG. 1). Generally, in the three-port evaporator 500, liquid flows into a liquid inlet 505 and into a core 510, defined by a primary wick 540, and fluid from the core 510 flows from a fluid outlet 512 to a ~~cold-biased~~ cold-biased reservoir (such as reservoir 155). The fluid and the core 510 are housed within a container 515 made of, for example, aluminum. In particular, fluid flowing from the liquid inlet 505 into the core 510 flows through a bayonet tube 520, into a liquid passage 521 that flows through and around the bayonet tube 520. Fluid can flow through a secondary wick 525 (such as secondary wick 145 of main evaporator 115) made of a wick material 530 and an annular artery 535. The wick material 530 separates the annular artery 535 from a first vapor passage 560. As power from the heat source ~~Qin~~ Q_{in} 116 is applied to the evaporator 500, liquid from the core 510 enters a primary wick 540 and evaporates, forming vapor that is free to flow along a second vapor passage 565 that includes one or more vapor grooves 545 and out a vapor outlet 550 into the vapor line 130. Vapor bubbles that form within first vapor passage 560 of the core 510 are swept out of the core 510 through the first vapor passage 560 and into the fluid outlet 512. As discussed above, vapor bubbles within the first vapor passage 560 may pass through the secondary wick 525 if the pore size of the secondary wick 525 is large enough to accommodate the vapor bubbles. Alternatively, or additionally, vapor bubbles within the first vapor passage 560 may pass through an opening of the secondary wick 525 formed at any suitable location along the secondary wick 525 to enter the liquid passage 521 or the fluid outlet 512.

Please amend the first full paragraph appearing on page 16, lines 20-31, as follows:

Referring to FIG. 6, in another implementation, the main evaporator 115 is designed as a four-port evaporator 600, which is a design described in U.S. ~~application Ser.~~ Patent Application Serial No. 09/896,561, filed June 29, 2001, now U.S. Patent No. 6,889,754, issued May 10, 2005. Briefly, and with emphasis on aspects that differ from the ~~three-port~~ three-port evaporator configuration, liquid flows into the evaporator 600 through a fluid inlet 605, through a bayonet tube 610, and into a core 615. The liquid within the core 615 enters a primary wick 620 and evaporates, forming vapor that is free to flow along vapor grooves 625 and out a vapor outlet 630 into the vapor line 130. A secondary wick 633 within the core 615 separates liquid within the core from vapor or bubbles in the core (that are produced when liquid in the core 615 heats). The liquid carrying bubbles formed within a first fluid passage 635 inside the secondary wick 633 flows out of a fluid outlet 640 and the vapor or bubbles formed within a vapor passage 642 positioned between the secondary wick 633 and the primary wick 620 flow out of a vapor outlet 645.

Please amend the first and second full paragraphs appearing on page 17, lines 1-19, as follows:

Referring also to FIG. 7, a heat transport system 700 is shown in which the main evaporator is a four-port evaporator 600. The heat transport system 700 includes one or more heat transfer systems 705 and a priming system 710 configured to convert fluid within the heat transfer systems 705 into a liquid to prime the heat transfer systems 705. The four-port evaporators 600 are coupled to one or more condensers 715 by a vapor line 720 and a fluid line 725. The priming system 710 includes a cold-biased reservoir 730 hydraulically and thermally connected to a priming evaporator 735.

Design considerations of the heat transport system 100 include startup of the main evaporator 115 from a supercritical state, management of parasitic heat leaks, heat conduction across the primary wick 140, ~~cold biasing~~ cold-biasing of the cold reservoir 155, and pressure containment at ambient temperatures that are greater than the critical temperature of the working fluid within the heat transfer system 105. To accommodate these design considerations, the body

or container (such as container 515) of the main evaporator 115 or secondary evaporator 150 can be made of extruded 6063 aluminum and the primary wicks 140 and/or 190 can be made of a fine-pored wick. In one implementation, the outer diameter of the main evaporator 115 or secondary evaporator 150 is approximately 0.625 ~~inches~~ inch and the length of the container is approximately 6 inches. The reservoir 155 may be cold-biased to an end panel of the ~~radiator~~ heat sink 165 using the aluminum shunt 170. Furthermore, a heater (such as a ~~kapton~~ KAPTON® heater) can be attached at a side of the reservoir 155.

Please amend the paragraph bridging pages 17 and 18 (page 17, line 28, through page 18, line 2), as follows:

In one implementation, the secondary condenser 122 and the secondary fluid line 160 are made of tubing having an OD of 0.25 ~~inches.~~ inch. The tubing is bonded to the panels of the heat sink 165 using, for example, epoxy. Each panel of the heat sink 165 is an ~~8×19 inch~~ 8×19-inch direct condensation, aluminum radiator that uses a {fraction (1/16)}-inch thick face sheet. ~~Kapton~~ KAPTON® heaters can be attached to the panels of the heat sink 165, near the condenser 120 to prevent inadvertent freezing of the working fluid. During operation, temperature sensors such as thermocouples can be used to monitor temperatures throughout the heat transport system 100.

Please amend the first, second, third and fourth full paragraphs appearing on page 18, lines 3-31, as follows:

The heat transport system 100 may be implemented in any circumstances where the critical temperature of the working fluid of the heat transfer system 105 is below the ambient temperature at which the heat transport system 100 is operating. The heat transport system 100 can be used to cool down components that require cryogenic cooling.

Referring to FIGS. 8A-8D, the heat transport system 100 may be implemented in a miniaturized cryogenic system 800. In the miniaturized system 800, the lines 125, 130, 160 are made of flexible material to permit coil configurations 805, which save space. The miniaturized system 800 can operate at -238°C using neon fluid. Power input ~~Qin~~ Q_{in} 116 is approximately

0.3 to 2.5 W. The miniaturized system 800 thermally couples a cryogenic component (or heat source that requires cryogenic cooling) 816 to a cryogenic cooling source such as a cryocooler 810 coupled to cool the condensers 120, 122.

The miniaturized system 800 reduces mass, increases flexibility, and provides thermal switching capability when compared with traditional ~~thermally-switchable,~~ thermally switchable vibration-isolated systems. Traditional ~~thermally-switchable,~~ thermally switchable vibration-isolated systems require two flexible conductive links (FCLs), a cryogenic thermal switch (CTSW), and a conduction bar (CB) that form a loop to transfer heat from the cryogenic component to the cryogenic cooling source. In the miniaturized system 800, thermal performance is enhanced because the number of mechanical interfaces is reduced. Heat conditions at mechanical interfaces account for a large percentage of heat gains within traditional ~~thermally-switchable,~~ thermally switchable vibration-isolated systems. The CB and two FCLs are replaced with the low-mass, flexible, thin-walled tubing used for the coil configurations 805 of the miniaturized system 800.

Moreover, the miniaturized system 800 can function ~~of~~ in a wide range of heat transport distances, which permits a configuration in which the cooling source (such as the cryocooler 810) is located remotely from the cryogenic component 816. The coil configurations 805 have a low mass and low surface area, thus reducing parasitic heat gains through the lines 125 and 160. The configuration of the cooling source 810 within the miniaturized system 800 facilitates integration and packaging of the miniaturized system 800 and reduces vibrations on the cooling source 810, which becomes particularly important in infrared sensor applications. In one implementation, the miniaturized system 800 was tested using neon, operating at ~~25-40K.~~ 25 to 40K.

Please amend the first, second and third full paragraphs appearing on page 19, lines 1-30, as follows:

Referring to FIGS. 9A-9C, the heat transport system 100 may be implemented in an adjustable mounted or ~~Gimbaled~~ gimbaled system 1005 in which the main evaporator 115 and a portion of the lines 125, 160, and 130 are mounted to rotate about an elevation axis ~~1020~~ within a range of ±45° and a portion of the lines 125, 160, and 130 are mounted to rotate about an

azimuth axis ~~1025~~ within a range of ±220°. The lines 125, 160, 130 are formed from thin-walled tubing and are coiled around each axis of rotation. The system 1005 thermally couples a cryogenic component (or heat source that requires cryogenic cooling) ~~1016~~ such as a sensor 1016 of a cryogenic telescope to a cryogenic cooling source 1010 such as a cryocooler ~~1010~~ coupled to cool the condensers 120, 122. The cooling source 1010 is located at a stationary spacecraft 1060, thus reducing mass at the cryogenic telescope. Motor torque for controlling rotation of the lines 125, 160, 130, power requirements of the system 1005, control requirements for the spacecraft 1060, and pointing accuracy for the sensor 1016 are improved. The ~~cryocooler~~ cooling source 1010 and the radiator or heat sink 165 can be moved from the sensor 1016, reducing vibration within the sensor 1016. In one implementation, the system 1005 was tested to operate within the range of ~~70-115K~~ 70 to 115 K when the working fluid is nitrogen.

The heat transfer system 105 may be used in medical applications, or in applications where equipment must be cooled to below-ambient temperatures. As another example, the heat transfer system 105 may be used to cool an infrared (IR) ~~sensor, which~~ sensor that operates at cryogenic temperatures to reduce ambient noise. The heat transfer system 105 may be used to cool a vending machine, which often houses items that preferably are chilled to ~~sub-ambient~~ sub-ambient temperatures. The heat transfer system 105 may be used to cool components such as a display or a hard drive of a computer, such as a laptop computer, handheld computer, or a desktop computer. The heat transfer system 105 can be used to cool one or more components in a transportation device such as an automobile or an airplane.

Other implementations are within the scope of the following claims. For example, the condenser 120 and heat sink 165 can be designed as an integral system, such as, ~~for example,~~ a radiator. Similarly, the secondary condenser 122 and heat sink 165 can be formed from a radiator. The heat sink 165 can be a passive heat sink (such as a radiator) or a cryocooler that actively cools the condensers 120, 122.

Please amend the second full paragraph appearing on page 20, lines 5-13, as follows:

Evaporators are integral components in two-phase heat transfer systems. For example, as shown above in FIGS. 5A and 5B, the evaporator 500 includes an evaporator body or

container 515 that is in contact with the primary wick 540 that surrounds the core 510. The core 510 defines a flow passage for the working fluid. The primary wick 540 is surrounded at its periphery by a plurality of peripheral flow channels or vapor grooves 545. The channels 545 collect vapor at the interface between the wick 540 and the evaporator body 515. The channels 545 are in contact with the vapor outlet 550 that feeds into the vapor line 130 that feeds into the condenser 120 to enable evacuation of the vapor formed within the main evaporator 115.

Please amend the paragraph bridging pages 20 and 21 (page 20, line 31, through page 21, line 2), as follows:

Referring to FIG. 10, an evaporator 1000 for a heat transfer system includes a heated wall 1005, a liquid barrier wall ~~1010,~~ 1011, a primary wick 1015 between the heated wall 1005 and the inner side of the liquid barrier wall ~~1010,~~ 1011, vapor removal channels 1020, and liquid flow channels 1025.

Please amend the first full paragraph appearing on page 21, lines 3-10, as follows:

The heated wall 1005 is in intimate contact with the primary wick 1015. The liquid barrier wall ~~1010~~ 1011 contains working fluid on an inner side of the liquid barrier wall ~~1010~~ 1011 such that the working fluid flows only along the inner side of the liquid barrier wall ~~1010.~~ 1011. The liquid barrier wall ~~1010~~ 1011 closes the evaporator's envelope and helps to organize and distribute the working fluid through the liquid flow channels 1025. The vapor removal channels 1020 are located at an interface between a vaporization surface 1017 of the primary wick 1015 and the heated wall 1005. The liquid flow channels 1025 are located between the liquid barrier wall ~~1010~~ 1011 and the primary wick 1015.

Please amend the third and fourth full paragraphs appearing on page 21, lines 15-26, as follows:

The vapor removal channels 1020 are designed to balance the hydraulic resistance of the vapor removal channels 1020 with the heat conduction through the heated wall 1005 into the

primary wick 1015. The vapor removal channels 1020 can be electro-etched, machined, or formed in a surface with any other convenient method.

The vapor removal channels 1020 are shown as grooves in the inner side of the heated wall 1005. However, the vapor removal channels 1020 can be designed and located in several different ways, depending on the design approach chosen. For example, according to other implementations, the vapor removal channels 1020 are grooved into ~~the~~ an outer surface of the primary wick 1015 or embedded into the primary wick 1015 such that they are under the surface of the primary wick 1015. The design of the vapor removal channels 1020 is selected to increase the ease and convenience of manufacturing and to closely approximate one or more of the following guidelines.

Please amend the paragraph bridging pages 21 and 22 (page 21, line 27, through page 22, line 2), as follows:

First, the hydraulic diameter of the vapor removal channels 1020 should be sufficient to handle a vapor flow generated on the vaporization surface 1017 of the primary wick 1015 without a significant pressure drop. Second, the surface of contact between the heated wall 1005 and the primary wick 1015 should be maximized to provide efficient heat transfer from the heat source to vaporization surface 1017 of the primary wick 1015. Third, a thickness 1030 of the heated wall 1005, which is in contact with the primary wick 1015, should be minimized. As the thickness 1030 increases, vaporization at the surface of the primary wick 1015 is reduced and transport of vapor through the vapor removal channels 1020 is reduced.

Please amend the first and second full paragraphs appearing on page 22, lines 3-8, as follows:

The evaporator 1000 can be assembled from separate parts. Alternatively, the evaporator 1000 can be made as a single part by in-situ sintering of the primary wick 1015 between two walls having special mandrels to form channels on both sides of the primary wick 1015.

The primary wick 1015 provides the vaporization surface 1017 and pumps or feeds the working fluid from the liquid flow channels 1025 to the vaporization surface 1017 of the primary wick 1015.

Please amend the fourth full paragraph appearing on page 22, lines 19-22, as follows:

The force that drives or pumps the working fluid of a heat transfer system is a temperature or pressure difference between ~~the~~ vapor and liquid sides of ~~the~~ a primary wick. The pressure difference is supported by the primary wick and it is maintained by proper management of the incoming working fluid thermal balance.

Please amend the paragraph bridging pages 22 and 23 (page 22, line 28, through page 23, line 2), as follows:

One method is an organized heat exchange between reservoir and the environment. For evaporators having a planar design, such as those often used for terrestrial applications, the heat transfer system includes heat exchange fins on the reservoir and/or on the liquid barrier wall ~~1010~~ 1011 of the evaporator 1000. The forces of natural convection on these fins provide subcooling and reduce stress on the condenser and the reservoir of the heat transfer system.

Please amend the third and fourth full paragraph appearing on page 23, lines 14-27, as follows:

As mentioned, it is important to obtain a proper balance between the heat leak into the liquid side of the evaporator and the pumping capabilities of the primary wick. This balancing process cannot be done independently from the optimization of the condenser, which provides subcooling, because the greater heat leak allowed in the design of the evaporator, the more subcooling needs to be produced in the condenser. The longer the condenser, the greater are the hydraulic losses in a fluid ~~lines,~~ line, which may require different wick material with better pumping capabilities.

In operation, as power from a heat source is applied to the evaporator 1000, liquid from the liquid flow channels 1025 enters the primary wick 1015 and evaporates, forming vapor that is

free to flow along the vapor removal channels 1020. Liquid flow into the evaporator 1000 is provided by the liquid flow channels 1025. The liquid flow channels 1025 supply the primary wick 1015 ~~with the~~ with enough liquid to replace liquid that is vaporized on the vapor side of the primary wick 1015 and to replace liquid that is vaporized on the liquid side of the primary wick 1015.

Please amend the paragraph bridging pages 23 and 24 (page 23, line 28, through page 24, line 2), as follows:

The evaporator 1000 may include a secondary wick 1040, which provides phase management on a liquid side of the evaporator 1000 and supports feeding of the primary wick 1015 in critical modes of operation (as discussed above). The secondary wick 1040 is formed between the liquid flow channels 1025 and the primary wick 1015. The secondary wick 1040 can be a mesh screen (as shown in the FIG. 10), or an advanced and complicated artery, or a slab wick structure. Additionally, the evaporator 1000 may include a vapor vent channel 1045 at an interface between the primary wick 1015 and the secondary wick 1040.

Please amend the first and second full paragraphs appearing on page 24, lines 3-12, as follows:

Heat conduction through the primary wick 1015 may initiate vaporization of the working fluid in a wrong ~~place on~~ place, on a liquid side of the evaporator 1000 near or within the liquid flow channels 1025. The vapor vent channel 1045 delivers the unwanted vapor away from the primary wick 1015 into the two-phase reservoir.

The fine pore structure of the primary wick 1015 can create a significant flow resistance for the liquid. Therefore, it is important to optimize the number, the geometry, and the design of the liquid flow channels 1025. The goal of this optimization is to support a uniform, or close to uniform, feeding flow to the vaporization surface 1017. Moreover, as the thickness 1019 of the primary wick 1015 is reduced, the liquid flow channels 1025 can be ~~space~~ spaced farther apart.

Please amend the fourth full paragraph appearing on page 24, lines 26-32, as follows:

Referring to FIGS. ~~10-13,~~ 11-13, an annular evaporator 1100 is formed by effectively rolling the planar evaporator 1000 such that the primary wick 1015 loops back into itself and forms an annular shape. The evaporator 1100 can be used in applications in which the heat sources have a cylindrical exterior profile, or in applications where the heat source can be shaped as a cylinder. The annular shape combines the strength of a cylinder for pressure containment and the curved interface surface for best possible contact with the ~~cylindrically-shaped~~ cylindrically shaped heat sources.

Please amend the first and second full paragraphs appearing on page 26, lines 4-20, as follows:

Referring also to FIGS. ~~14A-H,~~ 14A-14H, an annular evaporator 1400 is shown having a liquid inlet 1455 and a vapor outlet 1460. The annular evaporator 1400 includes a heated wall 1700 (FIGS. 14G, 14H, 15A, and 15B), a liquid barrier wall 1500 (FIGS. 14G, 14H, and ~~17A-D),~~ 17A-17D), a primary wick 1600 (FIGS. 14G, 14H, and ~~16A-D)~~ 16A-16D) positioned between the heated wall 1700 and the inner side of the liquid barrier wall 1500, vapor removal channels 1465 (FIGS. 14H, 15A, and 15B), and liquid flow channels 1505 (FIG. 14H). The annular evaporator 1400 also includes a ring 1800 (FIGS. 14G and ~~18A-D)~~ 18A-18D) that ensures spacing between the heated wall 1700 and the liquid barrier wall 1500 and a ring 1900 (FIGS. 14G, 14H, and ~~19A-D)~~ 19A-19D) at a base of the evaporator 1400 that provides support for the liquid barrier wall 1500 and the primary wick 1600. The heated wall 1700, the liquid barrier wall 1500, the ring 1800, the ring 1900, and the primary wick 1600 are preferably formed of stainless steel.

The upper portion of the evaporator 1400 (that is, above the primary wick 1600) includes an expansion volume 1470 (FIG. 14H). The liquid flow channels 1505, which are formed in the liquid barrier wall 1500, are fed by the liquid inlet 1455. The primary wick 1600 separates the liquid flow channels 1505 from the vapor removal channels 1465 that lead to the vapor outlet 1460 through a vapor annulus 1475 (FIG. 14H) formed in the ring 1900. The vapor removal channels 1465 may be photo-etched into the surface of the heated wall 1700.

Please amend the third and fourth full paragraphs appearing on page 27, lines 12-27, as follows:

The Stirling system 2000 is designed as a Free Piston Stirling Cooler (FPSC), such as Global Cooling's model M100B (Available from Global Cooling Manufacturing, 94 N. Columbus Rd., Athens, ~~Ohio).~~ OH). The FPSC 2000 includes a linear motor portion 2005 housing a linear motor (not shown) that receives an AC power input 2010. The FPSC 2000 includes a heat acceptor 2015, a regenerator 2020, and a heat rejecter 2025. The FPSC 2000 includes a balance mass 2030 coupled to the body of the linear motor within the linear motor portion 2005 to absorb vibrations during operation of the FPSC 2000. The FPSC 2000 also includes a charge port 2035. The FPSC 2000 includes internal components, such as those shown in the FPSC 2100 of FIG. 21.

The FPSC 2100 includes a linear motor 2105 housed within the linear motor portion 2110. The linear motor portion 2110 houses a piston 2115 that is coupled to flat springs 2120 at one end and a displacer 2125 at another end. The displacer 2125 couples to an expansion space 2130 and a compression space 2135 that form, respectively, cold and hot sides. The heat acceptor 2015 is mounted to the cold side of the expansion space 2130 and the heat rejector is mounted to the hot side of the compression space 2135. The FPSC 2100 also includes a balance mass 2140 coupled to the linear motor portion 2110 to absorb vibrations during operation of the FPSC 2100.

Please amend the paragraph bridging pages 27 and 28 (page 27, line 28, through page 28, line 3), as follows:

Referring also to FIG. 22, in one implementation, a FPSC 2200 includes heat rejector 2205 made of a copper sleeve and a heat acceptor 2210 ~~may~~ made of a copper sleeve. The heat rejector 2205 has an outer diameter (OD) of approximately 100 mm and a width of approximately 53 mm to provide a 166 cm^2 heat rejection surface capable of providing a flux of ~~6W/cm^2~~ 6 W/cm^2 when operating in a temperature range of ~~20-70 °C.~~ 20°C to 70°C. The heat acceptor 2210 has an OD of approximately 100 mm and a width of approximately 37 mm to

provide a 115 cm^2 heat accepting surface capable of providing a flux of ~~5.2W/cm²~~- 5.2 W/cm^2 in a temperature range of ~~-30-5 °C.~~ -30°C to 5°C.

Please amend the first and second full paragraphs appearing on page 28, lines 4-18, as follows:

Briefly, in operation an FPSC is filled with a coolant (such as, for example, ~~Helium~~ helium gas) that is shuttled back and forth by combined movements of the piston and the displacer. In an ideal system, thermal energy is rejected to the environment through the heat rejector while the coolant is compressed by the piston and thermal energy is extracted from the environment through the heat acceptor while the coolant expands.

Referring to FIG. 23, a thermodynamic system 2300 includes a cyclical heat exchange system such as a cyclical heat exchange system 2305 (for example, the systems 2000, 2100, 2200) and a heat transfer system 2310 thermally coupled to a portion 2315 of the cyclical heat exchange system 2305. The cyclical heat exchange system 2305 is cylindrical and the heat transfer system 2310 is shaped to surround the portion 2315 of the cyclical heat exchange system 2305 to reject heat from the portion 2315. In this implementation, the portion 2315 is the hot side (that is, the heat ~~rejector)of~~ rejector) of the cyclical heat exchange system 2305. The thermodynamic system 2300 also includes a fan 2320 positioned at the hot side of the cyclical heat exchange system 2305 to force air over a condenser of the heat transfer system 2310 and thus to provide additional convection cooling.

Please amend the fourth full paragraph appearing on page 28, lines 24-32, as follows:

Referring to FIG. 24, in another implementation, a thermodynamic system 2400 includes a cyclical heat exchange system such as a cyclical heat exchange system 2405 (for example, the systems 2000, 2100, 2200) and a heat transfer system 2410 thermally coupled to a hot side 2415 of the cyclical heat exchange system 2405. The thermodynamic system 2400 includes a heat transfer system 2420 thermally coupled to a cold side 2425 of the cyclical heat exchange system 2405. The thermodynamic system 2400 also includes fans 2430, 2435. The fan 2430 is positioned at the hot side 2415 of the thermodynamic system 2400 to force air through a

condenser of the heat transfer system 2410. The fan 2435 is positioned at the cold side 2425 of the thermodynamic system 2400 to force air through a condenser of the heat transfer system 2420.

Please amend the first, second and third full paragraphs appearing on page 29, lines 1-17, as follows:

Referring to FIG. 25, in one implementation, a thermodynamic system 2500 includes a heat transfer system 2505 coupled to a cyclical heat exchange system such as a cyclical heat exchange system 2510. The heat transfer system 2505 is used to cool a hot side 2515 of the cyclical heat exchange system 2510. The heat transfer system 2505 includes an annular evaporator 2520 that includes an expansion volume (or reservoir) 2525, a liquid return line 2530 providing fluid communication between liquid outlets 2535 of a condenser 2540 and ~~the~~ a liquid inlet of the evaporator 2520. The heat transfer system 2505 also includes a vapor line 2545 providing fluid communication between ~~the~~ a vapor outlet of the evaporator 2520 and vapor inlets 2550 of the condenser 2540.

The condenser 2540 is constructed from ~~smooth wall~~ smooth-wall tubing and is equipped with heat exchange fins 2555 or fin stock to intensify heat exchange on the outside of the tubing.

The evaporator 2520 includes a primary wick 2560 sandwiched between a heated wall 2565 and a liquid barrier wall 2570 and separating the liquid and the vapor. The liquid barrier wall 2570 is ~~cold biased~~ cold-biased by heat exchange fins 2575 formed along the outer surface of the heated wall 2565. The heat exchange fins 2575 provide subcooling for the reservoir 2525 and the entire liquid side of the evaporator 2520. The heat exchange fins 2575 of the evaporator 2520 may be designed separately from the heat exchange fins 2555 of the condenser 2540.

Please amend the paragraph bridging pages 29 and 30 (page 29, line 30, through page 30, line 4), as follows:

Initially, the liquid phase of the working fluid is collected in a lower part of the evaporator 2520, the liquid return line 2530, and the condenser 2540. The primary wick 2560 is wet because of ~~the~~ capillary forces. As soon as heat is applied (for example, the cyclical heat exchange system 2510 is turned on), the primary wick 2560 begins to generate vapor, which travels through ~~the~~ vapor removal channels (similar to vapor removal channels 1120 of evaporator 1100) of the evaporator 2520, through the vapor outlet of the evaporator 2520, and into the vapor line 2545.

Please amend the first full paragraph appearing on page 30, lines 5-14, as follows:

The vapor then enters the condenser 2540 at an upper part of the condenser 2540. The condenser 2540 condenses the vapor into liquid and the liquid is collected at a lower part of the condenser 2540. The liquid is pushed into the reservoir 2525 because of the pressure difference between the reservoir 2525 and the lower part of the condenser 2540. Liquid from the reservoir 2525 enters liquid flow channels of the evaporator 2520. The liquid flow channels of the evaporator 2520 are configured like the vapor removal channels 1125 of the evaporator 1100 and are properly sized and located to provide adequate liquid replacement for the liquid that vaporized. Capillary pressure created by the primary wick 2560 is sufficient to withstand the overall LHP pressure drop and to prevent vapor bubbles from travelling through the primary wick 2560 toward the liquid flow channels.

Please amend the fourth full paragraph appearing on page 30, lines 25-30, as follows:

The evaporator 2605 includes a heated wall 2700, a liquid barrier wall 2705, a primary wick 2710 positioned between the heated wall 2700 and ~~the~~ an inner side of the liquid barrier wall 2705, vapor removal channels 2715, and liquid flow channels 2720. The liquid barrier wall 2705 is coaxial with the primary wick 2710 and the heated wall 2700. The liquid flow channels 2720 are fed by a liquid return channel 2725 and the vapor removal channels 2715 feed into a vapor outlet 2730.

Please amend the first full paragraph appearing on page 31, lines 4-18, as follows:

In one implementation, the evaporator 2605 is approximately 2" tall and the expansion volume 2615 is approximately 1" in height. The evaporator 2605 and the expansion volume 2615 are wrapped around a portion of the cyclical heat exchange system 2610 having a 4" outer diameter. The vapor line 2620 has a radius of 1/8". The cyclical heat exchange system 2610 includes approximately 58 condenser channels 2625, with each condenser channel 2625 having a length of 2" and a radius of ~~0.012,"~~ 0.012", the channels 2625 being spread out such that the width of the condenser 2630 is approximate 40". The liquid return channel 2725 has a radius of {fraction (1/16)}". The heat exchanger 2800 (which includes the condenser 2630 and the fin stock 2640 is approximately 40" long and is wrapped into an inner and outer loop (see FIGS. 30, 33, and 34) to produce a cylindrical heat exchanger having an outer ~~diamter~~ diameter of approximately 8". The evaporator 2605 ~~have~~ has a cross-sectional width 2750 of approximately ~~1/8,"~~ 1/8", as defined by the heated wall 2700 and the liquid barrier wall 2705. The vapor removal channels 2715 have widths of approximately 0.020" and depths of approximately 0.020" and are separated from each other by approximately 0.020" to produce 25 channels per inch.

Please amend the third and fourth full paragraphs appearing on page 32, lines 15-32, as follows:

The vaporized working fluid exits the evaporator 3015 through the vapor outlet 3020 and enters a vapor line 3040 of the condenser 3010. The working fluid flows downward from the vapor line 3040, through channels 3045 of the condenser 3010, ~~to the~~ to a liquid return line 3050. As the working fluid flows through the channels 3045 of the condenser 3010 it loses heat, through the fins 3030 to the air passing between the fins, to change phase from vapor to liquid. Air that has passed through the fins 3030 of the condenser 3010 flows away through the exhaust channel 3035. Liquefied working fluid (and possibly some uncondensed vapor) flows from the liquid return line 3050 back into the evaporator 3015 through the liquid return port 3055.

Referring to FIGS. 33 and 34, a heat transport system 3300 surrounds a portion of a cyclical heat exchange system ~~3302,~~ 3302 that is surrounded, in turn, by exhaust channels 3305. The heat transport system 3300 includes an evaporator 3310 having an upper portion that surrounds the cyclical heat exchange system 3302. A vapor port 3315 connects the evaporator 3310 to a vapor line 3312 of a condenser 3320. The vapor line 3312 includes an outer region that circles around the evaporator 3310 and then doubles back on itself at junction 3325 to form an inner region that circles back around the evaporator 3310 in the opposite direction. The heat transport system 3300 also includes cooling fins 3330 on the condenser 3320.

Please amend the first through sixth full paragraphs appearing on page 33, lines 1-29, as follows:

The heat transport system 3300 also includes a liquid return port 3400 that provides a path for condensed working fluid from ~~the~~ a liquid line 3405 of the condenser 3320 to return to the evaporator 3310.

As mentioned above, the interface between the evaporator 3310 and the heat rejection surface of the cyclical heat exchange system 3302 may be implemented according one of several ~~alternate~~ alternative implementations.

Referring to FIG. 35, in one implementation, an evaporator 3500 slips over a heat rejection surface 3502 of a cyclical heat exchange system 3505. The evaporator 3500 includes a heated wall 3510, a liquid barrier wall 3515, and a wick 3520 sandwiched between the ~~walls~~ heated wall 3510 and the liquid barrier wall 3515. The wick 3520 is equipped with vapor channels 3525 and liquid flow channels 3530 are formed at the liquid barrier wall 3515 in simplified form for clarity.

The evaporator 3500 is slipped over the cyclical heat exchange system ~~3050~~ 3505 and may be held in place with the use of a clamp 3600 (shown in FIG. 36). To aid heat transfer, thermally conductive grease 3535 is disposed between the cyclical heat exchange system ~~3050~~ 3505 and heated wall 3510 of the evaporator 3500. In an ~~alternate~~ alternative implementation, the vapor channels 3525 are formed in the heated wall 3510 instead of in the wick 3520.

Referring to FIG. 37, in another implementation, an evaporator 3700 is fit over a heat rejection surface 3702 of a cyclical heat exchange system 3705 with an interference fit. The evaporator 3700 includes a heated wall 3710, a liquid barrier wall 3715, and a wick 3720 sandwiched between the ~~walls~~ heated wall 3710 and the liquid barrier wall 3715. The evaporator 3700 is sized to have an interference fit with the heat rejection surface 3702 of the cyclical heat exchange system 3705.

The evaporator 3700 is heated so that its inner diameter expands to permit it to slip over the unheated heat rejection surface 3702. As the evaporator 3700 cools, it contracts to fix onto the cyclical heat exchange system 3705 in an interference fit relationship. Because of the tightness of the fit, no thermally conductive grease is needed to enhance heat transfer. The wick 3720 is equipped with vapor channels 3725. In an ~~alternate~~ alternative implementation, the vapor channels are formed in the heated wall 3710 instead of in the wick 3720. Liquid flow channels 3730 are formed at the liquid barrier wall 3715 in a simplified form for clarity.

Please amend the first full paragraph appearing on page 34, lines 5-12, as follows:

The evaporator 3800 includes a wick 3820 and a liquid barrier wall 3815 formed about the modified heat rejection surface 3802, the wick 3820 and the liquid barrier wall 3815 being integrally bonded to the heat rejection surface 3802 to form ~~a~~ the sealed evaporator 3800. Liquid flow channels 3830 are portrayed in a simplified form for clarity. In this way, a hybrid cyclical heat exchange system with an integrated evaporator is formed. This integral construction provides enhanced thermal performance in comparison to the clamp-on construction and the interference fit construction because thermal resistance is reduced between the cyclical heat exchange system 3805 and the wick 3820 of the evaporator 3800.

Please amend the third and fourth full paragraphs appearing on page 34, lines 20-30, as follows:

As shown, at an air flow of 300 CFM, if the interface is a thermal grease interface, then the maximum amount of heat rejection would fall within a maximum heat rejection surface temperature 2907 (for example, ~~70 °C)~~ 70°C) with a heat exchange surface area 2910 (for

example, 100 ft^2). When the evaporator is constructed integrally with the portion by forming vapor channels directly in the heat rejection surface, that heat rejection surface would operate below the maximum heat rejection surface temperature of the thermal grease interface with significantly smaller heat exchange surface areas.

Referring to FIG. 39, a condenser 3900 is formed with fins 3905, which provide thermal communication between the air or the environment and a vapor line 3910 of the condenser 3900. The vapor line 3910 couples to a vapor outlet 3915 that connects ~~the an~~ an evaporator 3920 positioned within the condenser 3900.

Please amend the second full paragraph appearing on page 35, lines 15-23, as follows:

Referring to FIG. 46, a cross-sectional view of one side of a heat transfer system 4600 that is coupled to a cyclical heat exchange system 4605. This view shows relative dimensions that provide for particularly compact packaging of the heat transfer system. In this view, fins 4610 are portrayed as being 90 degrees out of phase for ease of illustration. To cool ~~the heat~~ heat rejection surface 4615 of the cyclical heat exchange system 4605 having a ~~4 inch~~ 4-inch diameter, the evaporator 4620 has a thickness of 0.25 inch and the radial thickness of the condenser is 1.75 inches. This provides on overall dimension for the packaging (the combination of the heat transfer system 4600 and the cyclical heat exchange system 4605 of 8 inches.